1.0

4.5
5.0
5.6
6.3

2.8

2.5

3.2

2.2

3.6

1.1

4.0

2.0

1.8

1.25

1.4

1.6

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FOREIGN TECHNOLOGY DIVISION

CALCULATION OF THE HYDRAULIC FRICTION OF THE
COOLING CHANNELS OF GAS-TURBINES BLADE

by

L. N. Odivanov

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MACHINE TRANSLATION

FTD-ID(RS)T-1827-77   27 Oct 1977

MICROFICHE NR: *FTD-77-C-001352*

CALCULATION OF THE HYDRAULIC FRICTION OF THE
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English pages: 27

Source: Trudy Kazanskogo Aviatsionnogo
          Institute, No. 128, 1971, pp. 3-13.

Country of origin: USSR
Translated by: Robert D. Hill
Requester: FTD/PDRS
Approved for public release; distribution
unlimited.

In the interest of economy and timeliness, the original
graphics have been merged with the computer output and
editing has been limited to that necessary for compre-
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Block	Italic	Transliteration	Block	Italic	Transliteration
А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
Ж ж	*Ж ж*	Zh, zh	Ц ц	*Ц ц*	Ts, ts
З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
К к	*К к*	K, k	Ъ ъ	*Ъ ъ*	"
Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

*ye initially, after vowels, and after ъ, ь; e elsewhere.
When written as ё in Russian, transliterate as yё or ё.

RUSSIAN AND ENGLISH TRIGONOMETRIC FUNCTIONS

Russian	English	Russian	English	Russian	English
sin	sin	sh	sinh	arc sh	$\sinh^{-1}$
cos	cos	ch	cosh	arc ch	$\cosh^{-1}$
tg	tan	th	tanh	arc th	$\tanh^{-1}$
ctg	cot	cth	coth	arc cth	$\coth^{-1}$
sec	sec	sch	sech	arc sch	sech^{-1}
cosec	csc	csch	csch	arc csch	csch^{-1}

Russian	English
rot	curl
lg	log

SUBJECT CODE 5147D

Page 3.

CALCULATION OF THE HYDRAULIC FRICTION OF THE COOLING CHANNELS OF
GAS-TURBINES BLADE

L. N. Odivanov.

Is given the derivation of the calculation formulas for
determining the hydraulic friction of the cooling channels of blades.
In formulas is considered the effect of the preheating of air, losses
from friction and centrifugal backwater on the capacity of cooling
channels.

For blades with radial (longitudinal) cooling channels are given
the calculation formulas, suitable for a programming on small ETsVM [
- digital computer]. For blades with transverse channels is given an
example of the calculation on the basis of the selection of the loss
factors in handbook. Calculation data satisfactorily coincided with

the experimental data of other authors.

In gas turbine construction ~~there~~ was received ~~the~~ wide acceptance of blade air-cooled. The diagrams of some of them are given in Fig. 1, 2.
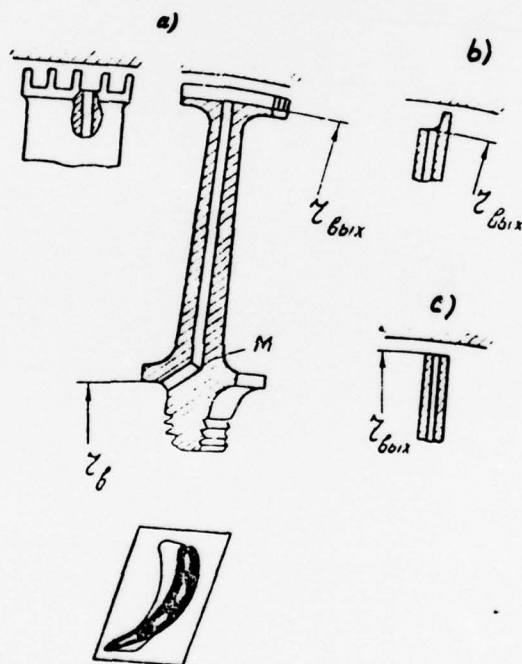
Fig. 1. Schematics of the cooled blades with the longitudinal channels.

Page 4.

~~In~~ During the ~~layout~~ *design* of cooling channels it is necessary correctly to estimate hydraulic losses. In the existing methods of calculation [1], etc. *there* is not examined the combined effect of some special ~~feature~~ peculiarities of air flow in the cooling channels of the blades: the effect of the preheating of air from the heat, ~~applied~~ *fed* through walls from gas; the heat, isolated from friction; the effect of centrifugal forces, etc. This leads to certain arbitrariness in the calculations, frequently ~~it~~ impedes the account of the specific conditions of the flow of coolant and the generalization of the loss factors.

In work [2] is experimentally investigated the effect of the factors indicated for a rotor with the blades, which have ~~within~~ *inside* deflector (Fig. 2). The clearance between the external wall and the deflector was maintain/withstood because of point punch-outs on deflector.

In works [3, 4] is investigated the ~~construction~~ *design* of blade with the transverse motion of air coolant, with its ejection in the area of trailing edge to the side of trough. Experimental data [4, 5],

etc. confirm the possibility of applying reference data [6] with some corrections.

In the present work is proposed the method of the calculation of hydraulic friction and determination of the flow rate of cooling air for blades with the longitudinal and cross flow of coolant with the assigned/~~proscribed~~ geometry of channels and parameters of air coolant.

Below is set forth the derivation of the calculation formulas on the basis of the equations of energy and continuity.
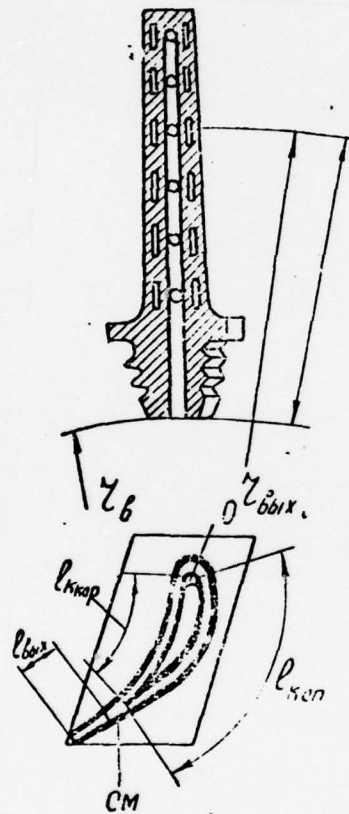
Fig. 2.
(Caption on next page.)

Diagram

Fig. 2. ~~Schematic~~ of ~~the~~ *a* cooled blade with transverse channels.

Page 5.

1. Equation of energy.

The equation of energy for air flow in cooling channels (in relative motion) is ~~record~~/written in the form:

$$c_p(T_{\text{н}} - T_{\text{вых}}) + Q + \frac{u_{\text{вых}}^2 - u_{\text{н}}^2}{2} - \frac{w_{\text{вых}}^2 - w_{\text{н}}^2}{2} = 0. \qquad (1)$$

Here Q - the heat, applied to air from the walls of the heated channel; *w* χ is ϕ relative speed; u - peripheral speed; T - the temperature of air in channels; c_p is the specific heat of air.

Subscripts

~~Indices~~ "в" and "вых" designate the parameters of air coolant at the duct inlet and at ~~output~~/~~yield~~ *outlet* from it respectively (Figs. 1, 2).

According to the equation of the 1st law of the thermodynamics

$$dL_r + dQ = c_p dT - \frac{dp}{\gamma}, \qquad (2)$$

where L_p is the work lost in friction; p and γ - the pressure and

air density.

In the process of the expansion of air its preheating through walls from gas, from friction and from centrifugal compression let us consider ~~through~~ *in terms of* the single average index *polytropic* ~~electromesons~~ n, i.e., let us accept $p/\tau^n = \text{const.}$ *Then integration of (2) will give*

$$Q = c_p(T_{max} - T_{\text{н}}) + \frac{n}{n-1} R T_{\text{н}}\left[1 - \left(\frac{p_{max}}{p_{\text{н}}}\right)^{\frac{n-1}{n}}\right] - \int_{\text{н}}^{\text{вых}} dL_r. \qquad (3)$$

It is known that

$$\int_{\text{в}}^{\text{вых}} dL_r = L_r = \frac{w^2_{\text{вых.n}} - w^2_{max}}{2}, \qquad (4)$$

where $w_{\text{вых n}}$ ~~is pitch~~ *the theoretical* speed without taking into account ~~of~~ hydraulic losses.

Taking into account expressions (3) and (4) equation (1) takes the form

$$w^2_{\text{вых.n}} = 2\frac{n}{n-1} R T_{\text{н}}\left[1 - \left(\frac{p_{max}}{p_{\text{н}}}\right)^{\frac{n-1}{n}}\right] + u^2_{max} - u^2_{\text{н}} + w^2_{\text{н}} \qquad (5)$$

Polytropic

Index ~~electromesons~~ it is possible to determine from the known relation~~ship~~/ratio

$$\frac{n-1}{n} = \frac{\lg \frac{T_{max}}{T_{к}}}{\lg \frac{p_{max}}{p_{в}}}, \tag{6}$$

and

~~и~~ outlet temperature from channel from the equation

$$T_{max} = T_{н} + \Delta T^* - \frac{w_{max}^2 - w_{в}^2}{2 \frac{k}{k-1} R}. \tag{7}$$

Page 6.

For determining w_{max} from value $w_{max.н}$ it is necessary to find the factor of hydraulic loss.

2. Factors of hydraulic loss.

For blades of the type a, b, c (Fig. 1) ~~of~~ duct loss*es* ~~they~~ are composed of the following forms: input, exit and loss from friction in channel. It is convenient to ~~relate~~ *refer* losses to real outlet velocity from channel, then $L_r = \xi_k \frac{w_{max}^2}{2}$ either according to expression (4) we

will obtain

$$w_{\text{вых}}^2 = \frac{w_{\text{вых.и}}^2}{1 + \xi_k}$$

or

$$w_{\text{вых}}^2 = \frac{2\frac{n}{n-1}RT_{\text{п}}\left[1 - \left(\frac{p_{\text{вых}}}{p_{\text{и}}}\right)^{\frac{n-1}{n}}\right] + u_{\text{вых}}^2 - u_{\text{в}}^2 + w_{\text{в}}^2}{1 + \xi_k} \,. \tag{8}$$

The total energy losses in channel, referred to the unit of mass, will be:

$$\xi_k \frac{w_{\text{вых}}^2}{2} = \Sigma \int_l \frac{w^{2}\zeta}{2D_{\text{г}}}\,dl + \Sigma \frac{\xi_{\text{м}} w_{\text{м}}^2}{2}\,.$$

Hence

$$\xi_k = \frac{1}{w_{\text{вых}}^2}\,\Sigma \int_l \frac{w^{2}\zeta}{D_{\text{г}}}\,dl + \Sigma \frac{\xi_{\text{м}} w_{\text{м}}^2}{w_{\text{вых}}^2}\,, \tag{9}$$

where $\xi_{\text{м}}$ is factor of local loss; ζ - the drag coefficient of the sections of channel; $D_{\text{г}}$ - the hydraulic diameter of the sections of channel; l is length of channel.

If we approximately accept a linear change in the losses from
the wall friction on the section of channel, then the first term of
expression (9) will take the form:

$$\xi = \frac{\zeta_{\text{н}} l}{2D_{\text{г.н}}}\left(\frac{w_{\text{н}}}{w_{\text{макс}}}\right)^2 + \frac{\zeta_{\text{макс}} l}{2D_{\text{г.вых}}}.$$

Speed change on section from output/yield to the inflection
point of channel is negligibly little, i.e., we assume that can be accepted

$$w_{\text{м}} = w_{\text{н}}.$$

Page 7.

Taking this into account the formula for determining the coefficient
of the losses of the channels, depicted on Fig. 1, will take the
form:

$$\xi_k = \frac{\zeta_{\text{вых}} l}{2D_{\text{г.макс}}} + \left(\frac{w_{\text{н}}}{w_{\text{макс}}}\right)^2\left(\xi_{\text{м}} + \frac{\zeta_{\text{в}} l}{2D_{\text{г.н}}}\right). \tag{10}$$

The velocity ratio is determined from the equation of the
continuity

$$\frac{w_{\text{н}}}{w_{\text{вых}}} = \frac{p_{\text{вых}} F_{\text{макс}} T_{\text{н}}}{p_{\text{н}} F_{\text{н}} T_{\text{макс}}}. \tag{11}$$

From equations (6), (7), (8), (10), (11) we is determined $w_{\text{макс}}$ and $T_{\text{макс}}$.
In this case pressure at output/yield from holes $p_{\text{макс}}$ can be accepted
approximately [1] that which is being changing according to linear law
from the gas pressure on periphery $p_{1\text{н}}$ at inlet entrance (into lattice) to $p_{2\text{н}}$
- at output/yield.

(FOOTNOTE) 1. More precise according to data [7] p_{max} a little

increases

~~flow/rises~~ with a decrease in the clearance δ and with an increase in

the dynamic head of air coolant. (ENDFOOTNOTE.)

For the blades, depicted on Fig. 2, are characteristic still

cther forms of ~~the~~ losses whose coefficients are enumerated below:

the coefficient, which evaluates loss from air distribution in

central channel ξ_p;

coefficients, the evaluating ~~entry~~ *at inlet* loss into hole 0 (Fig. 2),

losses ~~for~~ *from* shock of flow into wall (internal surface of the spout of

blade) and from the rotation of flcw in the direction of transverse

channels ξ_u and ξ_v;

the coefficient, which considers losses from the mixing of flows

upon their *encounter* ~~meeting~~ and from sudden exparsion *at outlet* ~~on output/yield~~ from the

channels of back and trough ξ_{cM}.

Are examined below recommendations *on* ~~to to~~ the selection of the

loss factors.

On the periphery of blades (Fig. 1b, c) the air flow at
output/~~yield~~ from hole depends on the effect of the flow of gas; the
mechanism of interaction is very complex; therefore the coefficient
of *outlet* ~~leaving~~ losses $\xi_{\text{вых}}$ can be determined experimentally. Tentative
calculations show that $\xi_{\text{вых}} = 0.3 \div 1$ (the upper limit corresponds to small
radial gap δ). For the blades, shown in Fig. 2, $\xi_{\text{вых}}$ it is possible to
determine by Borda-Carnot's formula: $\xi_{\text{вых}} = \left(1 - \dfrac{F_{\text{max}}}{F_{\text{n}}}\right)^2$.

Page 8.

In works [3, 4] the total losses $\xi_{\text{n}} + \xi_{\text{y}}$ are determined
experimentally; their value *is* ∧more at the smaller relative clearances $\bar{h}$
= h/d between the leading edge of the deflector and the internal
surface of blade. According to experiments [3] $\xi_{\text{n}} + \xi_{\text{y}} = 4 \div 5$ with $\bar{h} \approx$
1, while according to the data [4] 0.7-1.7 *when* ~~with~~ $\bar{h} \approx 2$.

The satisfactory agreement of these data is obtained, if ~~they are~~ *they*
are selected on graphs for circular exhaust shaft/~~mines~~ with hood,
upon consideration of additional entry loss into hole (account of the
evenness of entrance) [1, 6]. The coefficients of losses ξ_{n} at the
inlet ~~entrance~~ into cooling channels, $\xi_{p}, \xi_{\text{cм}}$ can be found from handbook [6].

The values of drag coefficients ζ_{R} ~~the~~ transverse channels,

obtained in experiments [4], satisfactorily coincide with ~~calculated~~ values

[6]. The drag coefficients of radial channels ζ_{H} and ζ_{max} are

determined from formulas for ducts. One must take into account that

the friction of the rotating radial channels is more than ~~being~~

~~unrotated~~ nonrotating ones. According to experiments [5] excess ~~composes~~ consists 15o/o.

These coefficients of losses and recommendation regarding their

selection require further refinement (especially for rotor blades)

with the accumulation of experimental data.

3. Determination of the preheating of coolant.

$$c_p dT^* == dQ + udu. \qquad (12)$$

The equation of energy in the complete parameters, the

differential form and in relative motion has the form:

The quantity of heat, applied to coolant (with flow rate per

second G) through walls, on the elementary section of the length

cf channel dz is equal to:

$$dQ_0 = G dQ - k_0(T_r^* - T^*)\,d\bar{z},$$

where

$$k_0 = \cfrac{l}{\cfrac{1}{\alpha_{\scriptscriptstyle H}\,\Pi_{\scriptscriptstyle H}} + \cfrac{\delta}{\lambda_{\scriptscriptstyle M}\,\Pi_{cp}} + \cfrac{1}{\sigma_{\scriptscriptstyle l}\,\Pi_{\scriptscriptstyle l}}} ;$$

$$\bar{z} = \frac{z}{l} ;$$

z - coordinate in the direction of a radius; δ is wall thickness; α - heat-transfer coefficient; $\lambda_{\scriptscriptstyle M}$ - the thermal conductivity of wall; Π - the perimeters of the ~~cross~~ section ~~cut~~ of wall.

By converting equation (12), we will obtain

$$dT^* = \left[k_0\,\frac{(T_r^* - T^*)}{G} + (\omega l)^2\,(\bar{r} + \bar{z}) \right] \frac{d\bar{z}}{c_p} .$$

Page 9.

At the average/~~mean~~ value of quantities k_0, T_r^* the definite integral of this equation in ~~range~~ *interval* from 0 to $\bar{z}$ takes the form:

$$\Delta T^* = \left[T_l^* - T_{\scriptscriptstyle H}^* + \frac{\omega^2 l^2}{c_p k_1}\left(\bar{r}_{\scriptscriptstyle H} - \frac{1}{k_1} \right) \right]\left(1 - e^{-k_1 \bar{z}} \right) + \frac{\omega^2 l^2 \bar{z}}{c_p k_1}, \quad (13)^{*)}$$

where $\bar{r}_{\scriptscriptstyle H} = \dfrac{r_{\scriptscriptstyle H}}{l}$, $r_{\scriptscriptstyle H}$ -   radius of the ~~entrance~~ *inlet* of hole; $T_{\scriptscriptstyle H}^*$ - the

stagnation temperature of coolant at the ~~entrance~~ *inlet* in relative motion;

$$k_1 = \frac{k_0}{Gc_p}.$$

(FOOTNOTE) 1. In book [8] is given the formula for determining the preheating of coolant in nozzle blades (ω = 0). ENDFOOTNOTE.

4. Determination of the ~~consumption~~ *flow rate* of coolant.

Outlet velocity from hole is ~~located through~~ *found from* converted expression (8)

$$w_{вых}^2 = \frac{\cdot\, 2\frac{n}{n-1}R(T_в - T_{вых}) + u_{вых}^2 - u_в^2 + \omega_в^2\left(1 - \xi_м - \frac{l_в}{2D_{г,в}}\right)}{1 + \frac{l_{вых}}{2D_{г,вых}}}. \tag{14}$$

The drag coefficient of channel with technical roughness with Re numbers > 2300 it is possible to determine from the general-purpose formula of Colbrook and White [6] taking into account an increase by 15o/o in the data [5] ~~during~~ *with* the rotation

$$\zeta = \frac{1,15}{\left[-2\lg\left(\frac{2,51}{Re\sqrt{\zeta}} + \frac{\Delta}{3,7D_г}\right)\right]^2}, \tag{15}$$

where Δ is roughness.

Moreover Re it is convenient to calculate according to the formula

$$\mathrm{Re} = \frac{4G}{\Pi \eta},\qquad(16)$$

where η is dynamic viscosity of coolant.

For determining $w_{\text{н}}$ they are assigned at first approximately by the ~~consumption~~ flow rate of coolant and then they find appropriate $p_{\text{н}}$, $T_{\text{н}}$ and $w_{\text{н}}$ through the following equations:

$$p_{\text{н}} = p_{\text{н}}^{\bullet}\left[1 - \frac{w_{\text{н}}^2(1 + \xi_{\text{н}})}{2\frac{k}{k-1}RT_{\text{н}}^{\bullet}}\right]^{\frac{k}{k-1}};\qquad(17)$$

$$T_{\text{в}} = T_{\text{в}}^{\bullet} - \frac{w_{\text{н}}^2}{\frac{2k}{k-1}R};\qquad(18)$$

$$w_{\text{в}} = \frac{GRT_{\text{н}}}{p_{\text{н}}F_{\text{в}}}.\qquad(19)$$

Page 10.

The calculation in channel is performed according to sections i, i. At output/yield from which from equations (6), (7), (13), (14) and the continuity

$$p_{\text{вых } i} = \left(\frac{GRT}{wF}\right)_{\text{вых } i} \qquad (20)$$

by the method successive approximations there they are determined $p_{\text{вых } i}$, $w_{\text{вых } i}$, $T_{\text{вых } i}$.

At output/yield from the last/latter section there it is calculated

$$p_{\text{max } p} = p_{\text{max } i} \left(1 - \frac{z_{\text{max}} w^2_{\text{max } i}}{2 \frac{k}{k-1} R T^*_{\text{max } i}} \right)^{\frac{k}{k-1}}.$$

The entire calculation is fulfilled also by the method of
~~approach~~/approximations G before agreement of $p_{\text{вых } p}$ with p_{max}
assigned~~/prescribed~~

The calculation is programmed for computation on by ETsVM "Nairi" ~~press~~.

The calculations show that the correctly approximate equality

$$\frac{n}{n-1} T_{\text{н}} \left[1 - \left(\frac{p_{\text{вых}}}{p_{\text{в}}} \right)^{\frac{n-1}{n}} \right] \approx$$

$$\approx \frac{k}{k-1} (T_{\text{в}} + 0{,}5 \Delta T^*) \left[1 - \left(\frac{p_{\text{вых}}}{p_{\text{в}}} \right) \right]^{\frac{k-1}{k}}, \qquad (21)$$

which is used below.

For the blades, depicted on Fig. 2, due to the presence of many
different forms of friction in coolant passage the calculation to

conveniently ^to^ conduct ~~consecutively~~ *sequentially* totaling hydraulic losses by sections along channel.

If we write equation (5) in stagnation parameters and ~~to~~ consider relation~~ship/ratio~~ (21), then **we will obtain** expression for determining the theoretical total pressure p_n^{*} on the appropriate radius of the blade

$$p_{u}^{*} = p_{u}^{*} \left[1 + \frac{u^2 - u_n^2}{2\dfrac{k}{k-1} R (T_n \div 0{,}5 \Delta T^{*})} \right]^{\frac{k}{k-1}}, \qquad (22)$$

where ΔT^* - the preheating of air on section from the entrance into radial channel before its distribution into transverse channels.

Page 11.

The temperature of stagnation was determined from converted expression (7)

$$T^* = T_n^{*} + \Delta T^*. \qquad (23)$$

The actual the total pressure at the output ~~yield on of~~ *outlet from* each section will be

$$p_{max}^{*} = p_{u}^{*} - \Delta p^*, \qquad (24)$$

where $\Delta p^* = \xi \dfrac{\gamma w^2}{2}.$

In handbook [6] the coefficients of losses ξ are given on the basis of experiments with small pressure differentials under isothermal conditions. In long channels with ~~the large~~ great preheating of coolant proceeds a considerable change in the losses and an ~~incidence~~/drop in the total pressure from heat supply. Taking this into account total pitot loss (in the channels of constant cross/section) approximately [1] will be

$$\Delta p^* \approx \left(\frac{\xi_{B}}{2} - 1 \right) \frac{\gamma_{H} w_{B}^{2}}{2} + \left(\frac{\xi_{max}}{2} + 1 \right) \frac{\gamma_{max} w_{max}^{2}}{2}$$

(for channels $\xi = \zeta \frac{l}{D_{r}}$).

FOOTNOTE [1]. The more precise value of the total pressure at output/~~yield~~ from section with the changing cross section can be found from equations (14) and (20). With the taken ~~consumption of~~ flow rate G is determined at first p_{max} and then $\dot{r}_{max}$. ENDFOOTNOTE.

For computation $\frac{\gamma w^{2}}{2}$ it is convenient to ~~apply~~ use gas-dynamic functions. With the/assigned ~~prescribed~~ flow rate in the first approximation, ~~consumption~~ is determined the function $q(\lambda) = \frac{G\sqrt{T^*}}{mFp^*}$ and from tables - λ.

Then *we find* $\frac{\gamma w^2}{2} = \frac{G}{2F} \lambda \sqrt{\frac{2k}{k+1}RT^*}$ and at the selected value ξ is determined Δp^*.

The calculation is fulfilled by the method *of* successive approximation, changing G before agreement in exit section $\lambda_{вых}$ from $\eta_з$ $= \frac{p_{вых}}{p_{вых}^*}$ ($p_{вых}^*$ *is* obtained by consecutive subtraction *of* Δp^* sections) and from the expression

$$y(\lambda_{вых}) = \frac{G\sqrt{T_{вых}^*}}{mF_{вых}\,p_{вых}}.$$

Page 12.

In this case pressure at output/~~yield~~ from hole can be taken as equal to the pressure of gas p_2 at output/~~yield~~ from lattice [1].

FOOTNOTE [1]. Work [2] shows the negligible effect of ejecting effect on trailing edge (from the flow of gas) on the ~~consumption~~ *flow rate* of air coolant. ENDFOOTNOTE.

The proposed calculation method makes it possible to determine coolant distribution according to span of the blade (Fig. 2).

Example of the calculation.

For *~~testing~~ checking* the method of calculation and selection of the loss factors was determined the flow rate of the air coolant through the channels of blades [2]. The loss factors were selected from handbook [6]:

$\xi_0 = 0.53 \div 0.56$ - for the holes of the rim of disk with transition to deflector;

$\xi_y = 1.58 \div 1.66$ - for a deflector (taking into account the separation of flow on holes on leading edge):

$\xi_{nx} + \xi_y = 1.1;$

$\xi_{m} = 4.76 \div 6.42$ - for the slots between deflector and wall of blade (was designed by formulas for lamellar-tubular radiators);

$\xi_{cm} = 1.14;$

$\xi_{l\,вих} = 0.141 \div 0.202;$

$\xi_{вих} = 0.77.$

In this case some geometric data, which are not given in work [2], are taken approximately from the figure of blade.

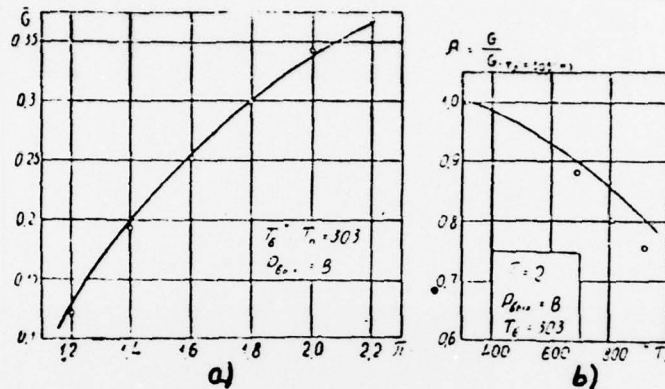Fig. 3. The dependence of the ~~given~~ *normalized* air flow rate on the ~~relation~~ *ratio* of pressures and temperature of the blades: — - the calculation; O - - according to experiments [3].

Page 13.

The preheating of air ΔT* from the sections of channel was determined from the equations of heat exchange and energy (13).

The results of the calculation in function $\bar{G} - \dfrac{G\sqrt{T_{\text{и}}^*}}{P_{\text{max}}\,F_{\text{max}}}$ on $\pi = \dfrac{P_{\text{G}}^*}{P_{\text{max}}}$ and from the temperature of blade $T_{\text{л}}$ were given in Fig. 3, where was visible the satisfactory convergence of calculated and experimental data [2].

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ORGANIZATION		MICROFICHE	ORGANIZATION		MICROFICHE
A205	DMATC	1	E053	AF/INAKA	1
A210	DMAAC	2	E017	AF/RDXTR-W	1
B344	DIA/RDS-3C	8	E404	AEDC	1
C043	USAMIIA	1	E408	AFWL	1
C509	BALLISTIC RES LABS	1	E410	ADTC	1
C510	AIR MOBILITY R&D	1	E413	ESD	2
	LAB/FIO			FTD	
C513	PICATINNY ARSENAL	1		CCN	1
C535	AVIATION SYS COMD	1		ETID	3
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C591	FSTC	5		NICD	5
C619	MIA REDSTONE	1			
D008	NISC	1			
H300	USAICE (USAREUR)	1			
P005	ERDA	1			
P055	CIA/CRS/ADD/SD	1			
NAVORDSTA (50L)		1			
NASA/KSI		1			
AFIT/LD		1			